A method for reducing noise of a high power combustion engine

TECHNICAL FIELD

The invention relates to sound attenuation of noise at the release of exhaust gases from a high power combustion engine.

BACKGROUND ART

An important factor to consider, when desiging an exhaust system connected to one or several high power combustion engines, is the noise in close vicinity to the outlet of the system. Examples of such a high power combustion engine are a diesel engine for a ship, for a power plant or for a train. By high power combustion engine is meant an engine with a total effect of more than 500 kW. A traditional method for providing a system of attenuators for sound attenuation comprises that such attenuators are built and supplied as one unit. Such a unit tends to become bulky.

A traditional system for sound attenuation of noise relating to an exhaust system of exhaust gases at a ship involves the use of a standardized system of attenuators, which are wrapped in walls of metals. Such a standardized system of attenuators may as such be called an attenuator and is a bulky unit with a large cross section area corresponding to a typical diameter of 2-4 meters. This means that in the case of a cruise ship, typically equipped with six diesel engines, the sound attenuators consume valuable space, which otherwise could be used for additional cabins.

It is known that one may reduce noise from an exhaust system of a high power combustion engine by use of different types of damping techniques. One way of reducing the noise is to arrange obstacles or steps of acoustic impedance to the progressing acoustic wave in the exhaust system channel. In this way, one prevents noise from propagating in the channel of the exhaust system. Such a type of sound attenuator is commonly known as a reactive attenuator. Such a reactive attenuator consumes no energy. There are two main principles to which such reactive attenuators works. A first type of reactive attenuator is a reflection attenuator, which comprises an increase of the cross-section area. This area increase gives rise to a reflection wave, which propagates in a direction opposite to the propagation of the sound. Such an obstacle may be regarded as a wall, in which the sound rebounds. A second type of reactive attenuator is a resonance attenuator, which influences the propagation of the sound in a channel. Such an obstacle functions as a pitfall, into which the progressing sound falls on its way towards the orifice. The sound-attenuation properties for a reactive attenuator are also dependent on where in the system the sound attenuator is placed.

Another type of attenuator is a resistive attenuator. One typical embodiment of a resistive attenuator is a round or square tube, the sides of which are coated with an absorbent or a porous medium of small coupled cavities. Such a sound attenuator intended for a ventilation system is described in the patent document GB 2,122,256. Another resistive attenuator intended for exhaust system is described in US 2,826,261. As absorbent, there is usually

used mineral wool or glass wool included some adhesive, which causes the absorbent to have a bonded structure. A gas-permeable surface layer, such as a perforated plate, may also protect the absorbent. Such a resistive attenuator will have a sound-attenuating property, which covers a wide frequency range and is dependant, besides on the thickness and rate of flow of the absorbent, also on the length and the inner area of the attenuator. The ratio of the absorbent thickness to the length of the acoustic waves, which are part of the sound is determining for the attenuation of lower frequencies. A satisfactory attenuation is achieved for sound frequencies at which the thickness of the absorbent is larger than a quarter of a wavelength of the sound. The sound attenuation properties then decrease drastically for sound of lower frequencies, which has a greater wavelength. Even when the ratio of the wavelength to absorbent thickness is about 1/8, the absorption is only half as great, and the ratio 1/16 it is only 20% of the absorption, which is obtained at the ratio 1/4. Since a certain absorption capacity remains, in many cases a sufficient absorption may be obtained by increasing the length of the total absorbent in the exhaust system. In addition, the cross-section area or diameter of the exhaust system is of importance for the sound attenuation obtained since the reduction in the upper frequency range of the sound decreases with increased cross-section area. Hence, a problem with such a resistive attenuator is that the absorbing layer must be thick to be able to absorb low frequencies. This means large volume. However, a larger total length of the attenuator may compensate a smaller absorbent thickness. This leads to an increased cost of the sound attenuation obtained. Another problem for an exhaust system is that

the pressure drop must be limited. This leads to a relatively large cross-section area of the system. The sound attenuation at the upper frequency range of the sound is thus reduced. It often appears that for traditional methods providing sound attenuation system properties, which are obtained in a laboratory, especially at low frequencies, are seldom obtained in practice. This leads to a great oversizing in order to insure sufficient sound attenuation.

The above-mentioned attenuators as well as other type of attenuators may be combined into an element. An example of such an element is described in US 4,371,054.

WO 98/27321 shows an example of a system with a reflective and a reactive attenuator. However, a remaining problem is to efficiently supply attenuators for elongated exhaust channels.

Methods for modeling noise in exhaust system may be based on four-pole theory, handling low frequencies, or based on power flow models, handling high frequencies. There is no known efficient method for modeling exhaust systems for large combustion engines with multiple attenuators and other components over the entire frequency range.

Yet another problem is during design of the system to efficiently model and try different combinations of elements with varying characteristics, and get an immediate result on the effect on sound attenuation, such as onboard a ship, depending on the combination of elements.

Another remaining problem is to arrange a sound attenuation system in a reliable manner, which not only meets noise requirements, but also insures that the sound attenuation system is not oversized nor overestimated from a design standpoint. Another problem is to during the design phase of a sound attenuation system also being able to minimize the pressure drop, since a large pressure drop reduces the efficiency of the combustion engine with or without a turbo charger.

A particular problem is design and assembly of a noise reduction system intended for a cruise ship, where demands on noise levels are rigorous, due to high demand on passenger comfort.

SUMMARY OF THE INVENTION

An object of the invention is to provide a method for supplying a system for sound attenuation of noise relating to an exhaust system of exhaust gases from a high power combustion engine, where elements intended for sound attenuation are, more efficiently and accurately than with known methods, modeled in a computing device, and the method does not have the above mentioned disadvantages. The method shall enable that an estimated noise level is calculated with high accuracy, not only in a band of low frequencies and in a band of high frequencies, but also in a band of intermediate frequencies. Hence, the method enables estimation of noise and attenuating effect in the whole frequency area. A system for sound attenuation according the invention shall enable the exhaust system to be supplied with attenuators which diameters are smaller compared with a traditional system for sound attenuation. A method according to the inven-

tion shall provide a system for sound attenuation where attenuator elements are placed along the exhaust system of exhaust gases, wherein such a system consumes less space and has less weight compared with exhaust system fitted with sound attenuators according to previously described traditional methods. Such a traditional method comprises the use of a bulky container of a system of attenuators. Further, the method according to the invention shall enable a user to quickly perform analysis of different design alternatives. The method enable that requirements on noise levels in close vicinity of the outlet of the exhaust system is met. The method shall enable to meet such requirements in a frequency range 25 Hz to 10 kHz. Further, the method shall provide less pressure drop than traditional methods.

The above mentioned object is met by a method according to claim 1.

The inventors have found that a particular suitable element to be used in the method, hence in the provided system for sound attenuation is an element where each element comprises a first reactive part, a resistive part and a second reactive part. An alternative name to such an element is a triple or a triple attenuator.

Another object of the invention is to provide a computer program capable of executing any of the steps according to the above-mentioned method.

Yet another object of the invention is to provide an exhaust system with a sound reduction system supplied according to the above-mentioned method.

It should be understood that the figures and description of embodiments are merely examples of possible embodiments and should not limit the underlying inventive idea.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will be described in more detail in connection with the enclosed schematic drawings.

Figure 1 shows an overview of a ship with an exhaust system and a position to which a desired noise level is associated.

Figure 2 shows a simplified flow diagram of a method according to the invention, a computing device and a user, such as an engineer, of the device.

Figure 3 shows an overview of an attenuator element, the characteristics of which are used in the modeling step of the method. That type of element is modeled at a plurality of positions in the exhaust system.

Figure 4 shows an example of an exhaust system with sound attenuation according to the invention.

Figure 5 is a schematic drawing of a display comprising entries to a number of functions, which relates to the invention.

Figure 6 is an overview, which shows that a contribution to an attenuating effect of an attenuation element is achieved in a low frequency area, an intermediate frequency area and an upper frequency area.

Figure 7 shows a simplified graph of a calculated attenuating effect of an attenuation element 20 in the low frequency area.

Figure 8 shows a simplified graph of an attenuation effect in the low frequency area of a sound reduction system comprising attenuation elements supplied according to the invention.

DETAILED DESCRIPTION OF THE INVENTION

Figure 1 shows an example of an exhaust system 1 for exhaust gases from a high power combustion engine, such as a diesel engine 4 for a ship 2. A position 3 is associated to a desired noise level. Such a position 3 is typically in close vicinity to the outlet of the exhaust system. A desired noise level is typically defined as an A-weighted value. Desired values are typically in the range of 60 dBA - 70 dBA.

As Figure 1 shows, the exhaust system comprises a number of different units. Examples of such units are a turbo charger, a boiler 5 or a heat exchanger 5. A heat exchanger is a common unit, in which part of the surplus heating of the hot gas is taken out for heating water or oil. In an embodiment an exhaust system where attenuator elements have been designed according to the invention, the attenuators 20 are positioned after the boiler 5. In an alternative embodiment, attenuators may also be positioned before the heat exchanger or the boiler.

Figure 2 shows a simplified flow diagram of a method according to the invention. Further, figure 2 indicates

that the steps of adding elements 7, inserting individual attenuating devices 8 and the calculating step 9 is performed by means of a computing device 13. It should be understood that an embodiment the invention may also comprise the modeling of the non-attenuated exhaust system, including the orifice, straight pipes, curved pipes etc. As an alternative, modeling of the non-attenuated exhaust system may be performed in another computing environment, different from the computing device. Constraints and details of the non-attenuated system may be defined in blueprints or as CAD-drawings or similar. The adding step 7 of figure 2 involves the modeling of a plurality of attenuator elements 20 shown in figure 3. Such elements comprise a first reactive part 21, a resistive part 22 and a second reactive part 23. The inventors have found that the use of such elements in a real world exhaust system 1, based on the invention, results in increased accuracy between an estimated noise level and a measured noise level. It is beneficial to use two to four such elements. In a method according to the invention, these elements are typically the first sound attenuators to be added to a model of a real world exhaust system. The position of such element in the channel is often dependent on available space between curved pipes. The adding step 7 typically involves an interaction between a user 14 and the computing device 13. The user 14, such as a design engineer, enters parameters relating to the elements through a user interface. Examples on such parameters relating to the element 20 are shown in figure 3 and may be D1, D2, xi smooth and xi perforation. Such sound damping characteristics is described in further detail in the description below of figure 3.

Further, the inventors have found that the use of the elements 20 enables an efficient combination of the sound damping characteristics of such elements 20 with individual reactive and resistive devices, which together gives the desired noise level. The inserting step of figure 2, involves the insert of at least one individual attenuating device. A typical real world attenuated system, such as for a cruise ship, based on the invention comprise at least four such individual devices. Such an individual device has its main damping effect in the low frequency area or in the upper frequency area. Such an individual device with damping in the low frequency area is of similar construction as part 21 or 23 of the attenuation element 20. Such an individual device with damping in the high frequency area is of similar construction as part 22 of the attenuation element 20. Hence, typically such an individual device is either a reactive or a resistive device. Further, during a repeat of the inserting step 8 additional individual attenuating devices may be added. Alternatively, to only adding a device in a repeat of the inserting step 8, acoustic characteristics of already added individual attenuating devices may be altered. In alternative embodiment the inserting step 8 may comprise that the position of already inserted elements or individual devices is altered. The change of position of reactive individual devices and elements has an impact on the total attenuation effect in the low frequency area. The attenuation effect in the high frequency area is dependent on the total length of resistive individual devices and the resistive parts 22 of the attenuation element. Further, the attenuation effect in the high frequency area is dependent on type of material r_0, total amount of

material, type of perforation x_i and other parameters of the resistive parts shown in figure 3.

The calculating step 9 comprises a calculation of an attenuating effect of the elements 20 and an attenuating effect of the single attenuating devices. It is beneficial to present the result of the calculation to the user of the computing device 13 as a total attenuating effect in a plurality of frequency bands. An aim is to make the attenuating effect at least equal to a needed attenuation in the real world exhaust system 1. The needed attenuation relates to the sound pressure level of the high power combustion engine. The needed attenuation may be calculated as the sound pressure level subtracted with the desired noise level. The calculating step 8 of figure 2 may comprise a calculation of an estimated noise level corresponding to a position in close vicinity of the outlet of the exhaust system.

In a preferred embodiment the calculation step 9 comprise that a contribution to an estimated attenuated effect comprise a band of frequencies corresponding to damping of intermediate frequencies related to an element 20. The damping is calculated by use of four-pole theory and by use of power flow models.

One system parameter, which one keeps constant in a preferred embodiment, is the sound pressure level of the high power combustion engine 4. The high power combustion engine 4 functions as a sound source to the exhaust system. The sound pressure level of the combustion engine is typically an anticipated sound pressure level and is supplied as input to the computing device as a sound

spectrum of band of frequencies. The manufacturer of the engine typically supplies or defines the sound spectrum. It is of particular importance that the sound spectrum is reliable and corresponds to the sound spectrum of the real world engine to be supplied as sound source of the real world exhaust system. In a preferred embodiment that sound spectrum is based on sound power, which is independent of distance from the sound source. However, a traditional method for supplying the sound spectrum of the engine involves the measurement of noise outside the engine tail pipe by means of a microphone. Such a method further involves that the measured sound spectrum is transformed by calculation to a sound power spectrum corresponding to the sound power of the combustion engine. Instead, it is according to the invention beneficial to measure the sound power as close to the inlet of the exhaust system 1 as possible. As an alternative, one may measure the sound power at the tail pipe of the combustion engine or a unit of the same type of engine before it is supplied as a sound source of the real world exhaust system. Since the desired noise level at the outlet 3 of the exhaust system typically is a maximum level, the sound power spectrum of the combustion engine 4 should correspond to full effect of the engine.

Another system parameter, which normally is kept constant, during the modeling and calculating step is the total length of exhaust channel. The total length, as well as other boundary conditions, is commonly indicated on blueprints or drawings.

It is beneficial that each element, individual attenuated device and other parts of the modeled exhaust system are

modeled as software objects or similar. In principle, the steps handled by means of a computing device may be implemented in software executable in any type of computing environment.

Figure 3 shows an overview of an attenuator element 20, the characteristics of which are used as parameters in the adding step 7 of the method. Figure 3 shows a cross section of such an attenuator element. That type of element is modeled at a plurality of positions in an exhaust system 47 according to Figure 4. Such an element comprises a first reactive part 21, a resistive part 22 and a second reactive part 23. Such an element may also be seen as an acoustic element, which suggests that the first reactive part 21, the resistive part 22 and the second reactive part 23 are not necessarily physically fitted together. The element is particular suitable due to the following. In an exhaust system 1, a sound field arises in the same way as in a room, which sound field is determined by the boundary conditions in the channel. There is a clearly expressed direction of movement of sound energy from the sound source 40 to the orifice 6, 46. The acoustic boundary conditions are thus determined by the properties of the limiting surfaces of the channel. Not least at the orifice are the acoustic boundary conditions complicated, since the very shape of the orifice, as well as the phenomenon that hot gas at a high pressure is thrown out into the air at normal temperature and normal atmospheric pressure, influence the sound generation. At the orifice, the progressing sound is subjected to strong reflection, whereby part of the sound energy passes in the opposite direction. The reflected sound gives rise to a sound field with standing

waves in the channel. In an unattenuated channel system, the sound field is determined almost exclusively by these reflection waves. Standing waves with pronounced nodes and great amplitudes are added to the generated sound field. By introducing attenuation in the channel system, the sound field becomes less accentuated. The inventors have found that by use of the element 20 it is possible to locally control the sound field generated in the channel. For low frequencies up to a cut on frequency an area increase causes a reflection wave where part of the progressing sound bounces back. In an attenuated elongated channel system 47, this means that, at such an area increase, a node in the sound field is located. The pipes of reactive parts 21 and 23 should be placed at a pressure maximum corresponding to the tuned frequency. The length L2 and L4 should correspond to approximately, but not exactly, a 1/4 of a wavelength for the tuned frequency. This in order for the pipes of the reactive parts 21 and 23 to take advantage of a reflective wave due to the area increase between part 21 and 22 respectively between part 22 and 23. This according to:

$$\lambda = c / f$$

where λ is the wavelength, c is the speed of sound and f is the frequency. One may note that the speed of sound depends on air temperature. Hence, in an exhaust system according to the invention it is important to consider changes in temperature along the channel 47.

Figure 4 shows an example of an exhaust system with sound attenuation according to the invention. 20a and 20b are two attenuating elements each having two reactive parts. The reactive parts typically have tuned frequencies between 65 Hz to 200 Hz. The elements 20a and 20b make

efficient use of available space and are cost effective to supply to the exhaust system. The exhaust system comprises an inlet 40, which is connected to a sound source such as a high power combustion engine 4. From an acoustic standpoint, the inlet 40 is considered an endless pipe. The exhaust system also comprises an outlet 46 or orifice, the shape and size of which have considerable influence on the sound in the channel. As the sound leaves the channel, it results in reflection waves. An exhaust system, such as shown in figure 4, often comprises a boiler 42 or heat exchanger. Such a boiler 42 has three main effects on the acoustic environment. The first effect is that the boiler 42 reduces the temperature of the exhaust gas and hence the speed of sound is different before and after the boiler 42. The second effect is that the boiler 42 may be seen as a boundary condition, similar to the orifice. A boiler 42, or rather an area increase/decrease in the boiler introduces distinct impedance. This makes it suitable to place single reactive devices as well as reactive parts of the attenuating element in relation to the boiler 42. It is an advantage to use such a relation to place the opening of a single attenuating device, such as 43 and 45 in figure 4, at an odd number of a quarter of a wavelength from a distinct impedance. The third main effect of the boiler 42 is that it has an attenuating effect, which in an embodiment of the invention is taken in account. It should be understood that figure 4 is schematic and a real world exhaust system comprise other parts than indicated in the figure such as bend pipes, flanges, connections to multiple engines etc.

Figure 5 is a schematic drawing of a display comprising entries to a number of functions 51-55 displayed by means of the computing device 13, which relates to the invention. The inventors have found that is an advantage to implement functions such as pre analysis 51, elements 52, system 53, source/termination 54 and post analysis 55. Such an elements function may comprise the definition of physical and acoustic characteristics of elements 20 as well as individual devices. Other examples of elements are a boiler, a heat exchanger, a pipe inlet, a pipe outlet or a flange. A system function may comprise that different elements are added or inserted to a model of an exhaust system 1. Many alternative definitions of functions are possible in an embodiment of the invention.

Figure 6 is an overview, which shows that a contribution to an attenuating effect in the intermediate frequencies 60 is achieved by use of the attenuator element 20. In an embodiment of the invention, an attenuating effect is calculated by use of band of frequencies. It is beneficial to use bands where each band corresponds to a tierce. Other distances between the bands are possible. Figure 6 shows that the attenuating effect of the element 20 in the low frequencies 61 is mainly achieved by the reactive parts 21 and 23. One should note that also resistive part 22 contributes to the attenuating effect in the low frequency area as a reflective attenuator. The resistive part 22 works as a reflective attenuator in both the direction of mass flow of exhaust gases, as well as a reflective attenuator in the other direction of reflective waves from other elements or, for instance, reflective waves from the orifice. The inventors have found that it is beneficial to start with the resistive

part 22 at the acoustic and physical design of a module. The length L3 should correspond to $\lambda/4$ based on a critical frequency to attenuate in the low frequency area. As an example: If a critical frequency is 125 Hz and the temperature of the exhaust gas at the position of the element is 350°C the speed of sound is calculated as:

$$c = 331{,}4 * \sqrt{\frac{1+350}{273}}$$

The speed of sound is 500 m/s which gives that for $\lambda/4$ L3 should be approximately 1 m. Part 21 and 23 should be tuned such that 125 Hz is the center frequency or close to the center frequency of the complete element in the low frequency area. It is in this case beneficial to tune part 21 to a frequency close to 110 Hz and part 23 close to 140 Hz. The above-mentioned calculation $\lambda/4$ in order to place the inlet of the pipes at pressure maximum gives that L2 of part 21 is 1.14 meters and L4 of part 23 is 0.89 meters.

Figure 7 shows a simplified graph of a calculated attenuating effect of an attenuation element 20 in the low frequency area. In the previous example, the center frequency 125 Hz corresponds to the centre 71 of the attenuation frequency band. The first reactive part 21 corresponds to the left 70 curve of an attenuation effect with a centre frequency close to 110 Hz. The second reactive part 22 corresponds to the right 72 curve of an attenuation effect with a centre frequency close to 125 Hz. It is beneficial to trim the acoustic performance of each attenuation element such that the total attenuation effect of the element, in the low frequency area, has flat top 73 corresponding to a certain damping measured in dB.

Figure 8 shows a simplified graph of the attenuation effect in the low frequency area of a sound reduction system comprising attenuation elements supplied according to the invention. Each of the curves 80, 81 and 82 corresponds to an attenuation element 20. The attenuation effect of each element 20 may result in damping effects with different amplitudes measured in dB, this in contrast to the figure, which shows a system with attenuation elements 20 with similar amplitude 80, 81 and 82 for its band of frequencies. One important advantage of a method according to the invention is that it enables a user to adapt the total attenuation effect of each of the attenuation elements 80, 81 and 82 to the corresponding sound effect of the high power combustion engine 4.

It should be understood that the elements and individual devices has a total attenuating effect greater than, if each attenuating effect would be added one by one to a total effect. The reason for that is that the elements and devices work together as a sound attenuation system. One acoustic effect that the invention makes use of is that one may introduce reflective waves by adding elements with resistive parts 21, that has a reflective character in the low frequency area, or individual resistive devices at suitable positions. Such suitable positions are odd numbers of 1/4 of desired wavelengths for attenuation.

The attenuation elements 20a and 20b, as well as individual devices 43, 45, are positioned in a model during the adding 7 and inserting step 8 such that the total attenuation effect of the attenuation elements 80, 81 and 82 and the attenuation effect of individual devices match

certain bands the frequency spectra of the corresponding sound effect of the high power combustion engine 4. A user 14 may try different combinations of individual devices and attenuation elements 20 during the repeating step 10 in order to achieve not only sufficient sound attenuation, but also to model the attenuation system such that it may be supplied to the exhaust system in a cost effective manner.

The attenuating effect in the high frequency 62 area of each element is mainly achieved by the resistive part 22. The calculation of the attenuating effect is made such that an estimated attenuated effect comprises a band of frequencies corresponding to intermediate frequencies 60 of an element 20. Such a contribution to frequencies of an element 20 is calculated by use of four-pole theory and by use of power flow models. Each element 20 contributes to attenuation in the low frequencies, intermediate frequencies and high frequencies. The inventors have found that it is beneficial to use a cut-on frequency to determine at what frequency the calculation should be based on four-pole theory or power flow models. The cut-on frequency depends on the cross-section area of transport system and the speed of sound $f_0(A,c)$. The cut-on frequency is typically calculated as:

$$f_0 \approx \frac{1{,}84 * c}{\pi * d}$$

where c is the speed of sound and d the diameter of the transport system at the element. According to Figure 6, below the cut-on frequency, the plane wave area, the attenuating effect of the element is mainly coming from the reactive parts 21 and 23.

In one embodiment the cut-on frequency determines that it is only the contribution from four-pole theory that is used in the calculating step 9 to add to the calculated attenuated effect below the cut-on frequency for each of the attenuation elements 20. And similar, in the same possible embodiment, the cut-on frequency determines that only the contribution from power flow models is used in the calculating step 9 to add to the calculated attenuated effect above the cut-on frequency.

Above the cut-on frequency, which is in the middle of the intermediate frequencies 60 of Figure 6, the attenuating effect of a real world attenuating element is mainly coming from the resistive part 22. It should be noted that the attenuated effect in the high frequency area of the resistive part 22 is not depending on the position of the element. Nor is the acoustic effect of the single attenuating devices with resistive character depending on position. However, it is beneficial to place at least some resistive attenuating devices before a possible boiler 42 or heat exchanger.

It is beneficial to position the attenuation elements 20 after a possible boiler 42 or heat exchanger. One reason is that commonly there are more straight pipes available after such a boiler 42. Another reason is that the temperature of the exhaust gases drops after a boiler 42 and heat exchanger. This means that the speed of sound is reduced after the boiler. This in turns means that to tune an element 20 for attenuation at a certain center frequency, such as 125 Hz, results in that the required total length of the element 20 is less after the boiler 42 than before the boiler.